REFRIGERANT FORCED-CONVECTION CONDENSATION INSIDE HORIZONTAL TUBES

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Report No. DSR 72591-71

American Society of Heating, Refrigerating and Air Conditioning Engineers
Contract No. ASHRAE RP 63

Engineering Projects Laboratory
Department of Mechanical Engineering
Massachusetts Institute of Technology
Cambridge, Massachusetts 02139

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TECHNICAL COMMITTEE 1.3

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ABSTRACT

High vapor velocity condensation inside a tube was studied theoretically. The heat transfer coefficients were calculated by the momentum and heat transfer analogy. The Von Karman universal velocity distribution was applied to the condensate flow. Pressure drop was calculated by the Lockhart-Martinelli method and the Zivi void fraction equation.

Experimental data was obtained for the mass velocities from 150,000 to 555,000 lbm/ft^2 hr for R-12 and R-22 condensing in a 0.493" I.D. 18 ft. long test section. The measured heat transfer coefficients agreed with the prediction within 10% except a few points in the very low quality region.

INTRODUCTION

When condensation takes place inside a horizontal tube with high vapor velocity, condensate flows in an annular shape on the tube wall and vapor flows in the core.

Many investigators have studied this subject both experimentally and analytically. Empirical correlations involving non-dimensional groups were not quite successful because the correlations did not include all the flow variables [1], [2], [5], [7], [12]. Carpenter and Colburn [6] considered only the laminar sublayer of condensate flow and derived an equation with an empirical constant. This method was modified by later investigators [3], [15]. For a small range of the Prandtl Number, this equation gives good agreement with empirical data. But the equation has no general applicability. Rohsenow, Webber and Ling [14] analyzed the liquid film on the vertical plate and the heat transfer coefficient was obtained by the heat and momentum transfer analogy. A similar approach appeared in later papers [8], [9]. This method will be developed further for the annular flow regime in this paper.

THEORY

Flow Model

For condensation inside a horizontal tube with high vapor velocities, annular flow is the predominent flow pattern and slug flow may appear at very low vapor qualities. Annular flow with a uniform liquid layer thickness around the circumference of a tube is assumed to exist in the parameter ranges of interest. The condensate accumulation at the bottom of a horizontal tube has a negligible effect except at very low vapor flow rates. At very high vapor flow rates entrainment of liquid droplet in the vapor may occur but this will be neglected in the following analysis.

Consider a length element of a tube, shown in Fig. 1. For the entire cross-section the momentum equation is

$$-\left(\frac{dP}{dz}\right)A_z - \tau_o S + A_z \frac{a}{g_o} [\alpha\rho_v + (1-\alpha)\rho_\ell] = \frac{1}{g_o}\frac{d}{dz}(U_v W_v + U_\ell W_\ell) \quad (1)$$

where a is an acceleration due to the external body force. Rearranging Eq. (1) yields

$$-\left(\frac{dP}{dz}\right) = \tau_o \frac{S}{A_z} - \frac{a}{g_o} [\alpha\rho_v + (1-\alpha)\rho_\ell] + \frac{1}{g_o A_z}\frac{d}{dz}(U_v W_v + U_\ell W_\ell) \quad (2)$$

The above equation shows that the total static pressure gradient is the sum of pressure gradients due to friction, gravity and momentum change.

$$\left(\frac{dP}{dz}\right) = \left(\frac{dP}{dz}\right)_f + \left(\frac{dP}{dz}\right)_g + \left(\frac{dP}{dz}\right)_m \quad (3)$$

Comparing Eq. (2) and (3),

$$\left(\frac{dP}{dz}\right)_f = -\tau_o \frac{S}{A_z} \quad (4)$$

$$\left(\frac{dP}{dz}\right)_g = \frac{a}{g_o} [\alpha\rho_v + (1-\alpha)\rho_\ell] \quad (5)$$

$$\left(\frac{dP}{dz}\right)_m = -\frac{1}{g_o A_z}\frac{d}{dz}(U_v W_v + U_\ell W_\ell) \quad (6)$$

The friction pressure drop was obtained by an approximation of the Lockhart-Martinelli method [11] as follows:

$$\left(\frac{dP}{dz}\right)_f \frac{g_o D}{G^2/\rho_v} = -\tau_o \frac{S}{A_z} \frac{g_o D}{G^2/\rho_v} = -0.09 \, \frac{GD}{\mu_v}^{-0.2} [x^{1.8} +$$

$$+ 5.7\left(\frac{\mu_\ell}{\mu_v}\right)^{0.0523} (1 - x)^{0.470} x^{1.33} \left(\frac{\rho_v}{\rho_\ell}\right)^{0.261}$$

$$+ 8.11\left(\frac{\mu_\ell}{\mu_v}\right)^{0.105} (1 - x)^{0.94} x^{0.86} \left(\frac{\rho_v}{\rho_\ell}\right)^{0.522}] \qquad (7)$$

This empirical approximation was developed by Soliman et al [15] who used this equation to calculate τ_v. Here it is used to calculate τ_o as Lockhart-Martinelli suggest.

The gravity term, Eq. (5), can be rewritten in the following form:

$$\left(\frac{dP}{dz}\right)_g \frac{g_o D}{G^2/\rho_v} = \frac{1}{Fr^2}[\frac{\rho_\ell}{\rho_v} - B\alpha] \qquad (8)$$

where

$$Fr^2 = \frac{(G/\rho_v)^2}{aD} \qquad (9)$$

is the Froude number based on the total flow and

$$B \equiv \frac{\rho_\ell - \rho_v}{\rho_v} \qquad (10)$$

is the buoyancy modulus. In the gravity field

$$a = g \sin \theta \qquad (11)$$

The Zivi equation for local void fraction [17] is recommended for use in Eq. (8), as in reference [15].

$$\alpha = \frac{1}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_v}{\rho_\ell}\right)^{2/3}} \qquad (12)$$

The momentum term, Eq. (6), can be written with Eq. (12) as follows:

$$\left(\frac{dP}{dz}\right)_m \frac{g_o D}{G^2/\rho_v} = - D\left(\frac{dx}{dz}\right)\left[2x + (1 - 2x)\left(\frac{\rho_v}{\rho_\ell}\right)^{1/3}\right.$$

$$\left. + (1 - 2x)\left(\frac{\rho_v}{\rho_\ell}\right)^{2/3} - 2(1 - x)\left(\frac{\rho_v}{\rho_\ell}\right)\right] \qquad (13)$$

where in Eq. (6)

$$W_v = GA_z x = A_z \alpha U_v \rho_v$$

$$W_\ell = GA_z(1-x) = A_z(1-\alpha)U_\ell \rho_\ell \qquad (14)$$

The momentum equation for the entire liquid layer element, Fig.2, is

$$-\left(\frac{dP}{dz}\right) A_{z_\ell} + \tau_v S_v - \tau_o S + \frac{a}{g_o} \rho_\ell A_{z_\ell}$$

$$= \frac{1}{g_o}\left[\frac{d(U_\ell W_\ell)}{dz} - U_i \frac{dW_\ell}{dz}\right] \qquad (15a)$$

or

$$\tau_o = F_o \frac{A_{z_\ell}}{S} + \tau_v \frac{S_v}{S} \qquad (15b)$$

where

$$F_o \equiv - \frac{dP}{dz} + \frac{a}{g_o} \rho_\ell - \frac{1}{g_o A_{z_\ell}}\left[\frac{d(U_\ell W_\ell)}{dz} - U_i \frac{dW_\ell}{dz}\right] \qquad (16)$$

Since for most of the tube length the liquid film is thin, a simple flat plate analysis will suffice for the heat transfer coefficient derivation.

The $A_{z_\ell}/S \simeq \delta$ and $S_v/S \simeq 1$; so Eq. (15b) becomes

$$\tau_o = F_o \delta + \tau_v \qquad (17)$$

The quantity F_o may be expressed in terms of x and α by substituting Eqs. (12) and (14) into Eq. (16). Further from the universal velocity, Eq. (B-1), distribution $(U_i/U_\ell) \equiv \beta$ may be obtained as a unique function of δ^+ as shown in Fig. 3. Then Eq. (17) becomes

$$F_o = -\left(\frac{dP}{dz}\right) + \frac{a}{g_o}\rho_\ell - \frac{G^2}{g_o\rho_v}\frac{dx}{dz}\left[\frac{1}{1-\alpha}\left(\frac{\rho_v}{\rho_\ell}\right)^{1/3} - \left(\frac{(1-x)(2-\beta)}{(1-\alpha)^2}\right)\left(\frac{\rho_v}{\rho_\ell}\right)\right] \tag{18}$$

where $\beta \equiv U_i/U_\ell$ is given by Fig. 3.

To determine the heat transfer coefficient it is assumed that the Karman momentum-heat transfer analogy analysis is applicable in the liquid layer. Then

$$\tau = \frac{\rho_\ell}{g_o}(\nu_\ell + \varepsilon_m)\frac{dv_z}{dy} \tag{19a}$$

$$q/A = \rho_\ell c_\ell (\alpha_\ell + \varepsilon_h)\frac{dT}{dy} \tag{19b}$$

The universal velocity distribution is used to determine dv_z/dy, ε_m is assumed equal to ε_h, and dT/dy and T(y) is determined by combining the above two equations. The procedure is identical with that presented in detail by Rohsenow, Webber and Ling [14] with two exceptions. In determining this temperature distribution, the momentum equation for the element (δ-y) of Fig. 2 was approximated as

$$\tau = F_o(\delta - y) + \tau_v$$

and no criterion for transition from laminar to turbulent flow in the film was used. The suggestion of Dukler [8] to integrate the equations and let the universal velocity distribution establish the flow regime was adopted. The details of the analysis are outlined in Appendix B. The results are

$$Nu_z \equiv \frac{h_z D}{k_\ell} = \frac{\rho_\ell c_\ell D u_\tau}{k_\ell F_2} \tag{21a}$$

or

$$St_z^* \equiv \frac{h_z}{\rho_\ell c_\ell u_\tau} = \frac{1}{F_2} \tag{21b}$$

where

$$u_\tau \equiv \sqrt{\frac{g_o \tau_o}{\rho_\ell}} \tag{22}$$

and

$$\text{for } 0 < \delta^+ < 5: \; F_2 = \delta^+ \, Pr \tag{23a}$$

$$\text{for } 5 < \delta^+ < 30: \; F_2 = 5Pr + 5\ell n[1 + Pr\,(\frac{\delta^+}{5} + 1)] \tag{23b}$$

$$\text{for } \delta^+ > 30: \; F_2 = 5Pr + 5\ell n(1 + 5Pr) + \frac{2.5}{\sqrt{1 + \frac{10}{Pr}\frac{M}{\delta^+}}} \; x$$

$$x \ln\left[\frac{2M-1 + \sqrt{1 + \frac{10}{Pr}\frac{M}{\delta^+}}}{2M-1 - \sqrt{1 + \frac{10}{Pr}\frac{M}{\delta^+}}} \cdot \frac{\frac{60}{\delta^+} M - 1 - \sqrt{1 + \frac{10}{Pr}\frac{M}{\delta^+}}}{\frac{60}{\delta^+} M - 1 + \sqrt{1 + \frac{10}{Pr}\frac{M}{\delta^+}}}\right] \tag{23c}$$

Here

$$M \equiv \frac{F_o \, \delta^+ \, \nu_\ell}{\tau_o \, u_\tau} \tag{24}$$

and

$$\delta^+ \equiv \delta \, u_\tau / \nu_\ell \tag{25}$$

Further Re_ℓ defined as

$$Re_\ell = \frac{(1-x)GD}{\mu_\ell} = \frac{4\Gamma}{\mu_\ell} = \frac{4}{\mu_\ell}\int_0^{\delta} \rho_\ell v_z dy = 4\int_0^{\delta^+} v_z^+ \, dy^+ \tag{26}$$

is evaluated from the velocity distribution Eq. (B-1) with the following results:

$$\delta^+ < 5 \qquad Re_\ell = 2(\delta^+)^2 \tag{27a}$$

$$5 < \delta^+ < 30 \qquad Re_\ell = 50 - 32.2\,\delta^+ + 20\delta^+ \ln\delta^+ \tag{27b}$$

$$\delta^+ > 30 \qquad Re_\ell = -256 + 12\delta^+ + 10\delta^+ \ln\delta^+ \tag{27c}$$

A plot of Re_ℓ vs δ^+ is shown in Fig. 4.

For any assumed magnitude of Pr, δ^+ and M, calculate Re_ℓ from Eq. (27), F_2 from Eq. (23) and St^* from Eq. (21b). Then curves of St^* vs Re_ℓ for various M can be constructed. Fig. 5 for Pr = 1 and 5 was drawn by this procedure.

The calculation procedure starts by dividing the tube length in increments of changes in quality x and for a given flow rate and fluid conditions calculate the increment of length required to accomplish this quality change. The calculation is a step-wise one requiring trial-and-error at each step. The procedure is outlined in a sample calculation in Appendix A.

Average Heat Transfer Coefficient

For the case of uniform wall temperature a mean heat transfer coefficient h_m may be defined by $h_m = (1/L)\int_0^L h_z dz$. Then

$$q = \Gamma_L \pi D h_{fg} = h_m \Delta T \, \pi DL \tag{28}$$

For an element of length dz

$$dq = \pi D\, h_{gh} d\Gamma = h_z \Delta T\, \pi D\, dz \tag{29}$$

Rearrange Eq. (29) and integrate

$$\int_o^{\Gamma_L} \frac{d\Gamma}{h_z} = \frac{\Delta T}{h_{fg}} \int_o^L dz = \frac{\Delta T}{h_{fg}} L = \frac{\Gamma_L}{h_m} \tag{30}$$

Then from Eq. (30) with Eq. (26)

$$\frac{1}{h_m} = \frac{1}{\Gamma_e} \int_o^{\Gamma_e} \frac{d\Gamma}{h_z} = \frac{1}{Re_{\ell,e}} \int_o^{Re_{\ell,e}} \frac{dRe_\ell}{h_z} \tag{31}$$

or since $Re_\ell \sim (1-x)$ from Eq. (26), this becomes

$$\frac{1}{h_m} = \frac{1}{x_e} \int_{x_e}^{1} \frac{dx}{h_z} \tag{32}$$

From the h_z calculated along the length, this length mean heat transfer coefficient may be calculated integrating with respect to quality as an alternative.

EXPERIMENT

The basic apparatus, schematically shown in Figure 6, consists of a closed-loop refrigerant flow circuit driven by a mechanical-sealed rotor pump. Upstream of the test section, an electrically heated boiler produces vapor, which passes through a flow meter and a throttle valve to the test section. Downstream of the test section, an after-condenser was provided to ensure fully condensed refrigerant at the pump inlet. The pump flow was set for any test run and flow rate in the test section was controlled by the by-pass loop. The pressure level was set by adjusting the heat input to the boiler and the throttle valve.

The test section itself is an annular shaped heat exchanger with refrigerant flowing through the inner tube and cooling water flowing in the outer annulus counter-currently. The 0.493 in. ID. smooth nickel tube test section was divided into six 3 ft-long sections. Each section has a separate cooling water circuit and the sections are connected smoothly with specially made stainless steel fittings in order not to disturb the condensate flow.

Each of the six sections except the third section from the inlet was separately and identically instrumented to give basic data on the condensing refrigerant. Two thermocouples are placed in the middle of the 3 ft section at the side; one at the outside of the condenser tube and the other one in the vapor at the center of the tube. Two differential thermocouples between the inlet and the outlet of the cooling water circuit are located in two different radial positions in order to detect any possible non-uniformity in temperature. On the third section, in addition to the above thermocouples, two more thermocouples are placed at the top and bottom of the tube wall to measure any circumferential variation of the wall temperature.

All the thermocouples were made of 0.005 in. O.D. nylon-sheathed copper and constantan wire.

Seven pressure taps were installed at every connection between the 3 ft sections for measurement of local pressure gradients.

All the loop except the part from the pump to the boiler was insulated with fiberglass. The heat transfer between the test section and the atmosphere was not measurable in a blanked off run with no vapor flow.

Data were taken after steady state had been attained for one hour in the system. The heat flux to the coolant was obtained from the coolant flow rate and the temperature change. The condensing wall temperature was determined from the outside tube wall temperature and the heat flux. All the measurements were done on one 3 ft section at a time from up-stream to down-stream. The coolant flow was regulated such that the wall temperatures were kept almost constant through the test section and the temperature change of the coolant was in the range of 1 to 3°F.

Heat balance was checked with total enthalpy change from the inlet of the test section to the outlet of the after-condenser. In most runs, except Run 1, the heat balance error was less than $\pm$ 6%.

The data for both R-12 and R-22 are tabulated in the Appendix. Pressure drop data was taken only for R-22. Figures 7, 8, 9 are samples of the plot of the data but are representative of all of the data. Additional plots of the data are presented in references [18] and [19].

DISCUSSION OF RESULTS

Since the theoretical analysis was based on the annular flow model, the results are applicable only to the case where annular flow is developed. To date no successful investigation has been made of condensation flow regimes. For gas and oil mixtures, a flow regime map was drawn by Baker [4], but it may not be applicable to two-phase flow with condensation. However, it surely gives an approximate view of the flow regime boundaries of condensation. Quandt [13] analyzed qualitatively the force field of gas-liquid flow. Still a quantitative figure of the flow regime boundaries cannot be obtained from an analysis. Therefore, until more reliable information about flow regimes of vapor-liquid flow with condensation is available, it is recommended that the Baker plot be used for determining probable flow regimes.

In most cases of practical forced-convection flow the regime appears to be annular except at the very low quality region. This analysis is not applicable to the very low quality region because the flow regime may be different and because the condensate film is so thick that the flat plate analysis is no longer valid for a tube. The present method is therefore not suggested for use when the vapor quality is less than 20%. A fared curve between the present correlation at $x = 0.20$ and McAdams equation for single phase flow ($x = 0$) will give useful information for the low quality region.

Entrainment of liquid in the vapor core was neglected in the analysis. Since thermal resistance is mainly offered by the laminar sublayer and the buffer layer, the entrainment effect is not significant when the condensate film thickness is larger than that of the high thermal resistance layers ($\delta^+ > 30$). However, as expected, the effect appears to be significant at the very high vapor quality region where a very thin film exists ($\delta^+ < 5$), as shown in some of the test runs. In a few runs at very high vapor flow rate when a considerable amount of entrainment was produced, the theory predicted lower values of h than those measured.

As the total flow rate decreases to low values, the thickness of the liquid film on the wall of a horizontal tube may be changed significantly. Even though the flow shape becomes an eccentric annulus, the analysis may give a good prediction because the heat transfer coefficient increases at the top and decreases at the bottom of the tube when this happens. However when truly stratefied flow exists another theory should be used.

The agreement with the present data is within 10% except for a few low quality points. In general predictions are slightly lower than the experimental data within the range of measurement accuracy, Figs. 7, 8, 10, and 11. The pressure drop measurements, Figs. 9 and 12, also show good agreement except for Run 8. It is interesting to note that at the upstream end of the condensing tube the predicted pressure gradient has a negative slope. However, the measurement shows the opposite trend. Except for Runs 5 and 8, the pressure drop of the first section is always higher than that of the other sections.

Other Comparisons

Figure 13 shows the present data plotted on coordinates suggested by Akers and Rosson [2]. The solid lines represent their recommended correlation equations. Practically all of the data fall well above this recommendation. A plot of this same data [18] on coordinates suggested by Brauser [5] shows an equally large scatter. It is not surprising that such scatter should exist. In Fig. 13 the h for a given ΔT and pressure is essentially a function of G_v independent of quality. For the same G_v the liquid layer thickness, which offers the primary heat transfer resistance, is greatly different at qualities of, say, 10% and 90%; hence h should be quite different.

The present data along with the data of Altman et al [3] was compared [18] with a prediction equation suggested by Boyko and Kruzhilin [20] and was found to scatter badly. In general, the data fell as much as 250% above and 100% below the suggested prediction.

Figure 5 shows a comparison of the present predicted results with the predictions of Carpenter and Colburn [6] and Kunz and Yerazunis [9]. The Carpenter-Colburn equation was derived considering only a laminar sub-layer and shows essentially no effect of the liquid Reynolds number. The coefficients were determined empirically for a limited range of data. The Kunz and Yerazunis study omitted the effect of D, gravitational effect and the momentum pressure gradient. Their result shows a discrepency from the present analysis at liquid Reynolds numbers above around 1000.

CONCLUSION

The proposed prediction method for forced convection condensation heat transfer involves a combination of and modification of several previous analyses, [14][15][8], and agrees with the present and other data to within $\pm$ 10% for refrigerants in the range of conditions commonly found in commercial refrigeration equipment.

ACKNOWLEDGEMENTS

This work was supported by Technical Committee 1.3 of ASHRAE. Valuable suggestions and advice were received during the course of the work from Professors A. E. Bergles and A. A. Sonin and to Mr. Donald Traviss.

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Nomenclature

A_z	cross section area ft^2
a	actual gravitational acceleration in the axial direction, $g \sin \theta$ ft/hr^2
B	buoyancy modulus, Eq. (10)
C_p	specific heat Btu/lbm °F
D	tube inner diameter ft
F_o	defined in Eq. (16) $lbf/ft^2/ft$
F_2	defined in Eq.(23a, b, c)
Fr	Froude number, Eq. (9)
g	gravitational acceleration ft/hr^2
g_o	constant, 4.17×10^8 lbm ft/lbf hr^2
G	total mass velocity lbm/ft^2
h_z	local heat transfer coefficient Btu/hr ft^2 °F
h_m	mean heat transfer coefficient Btu/hr ft^2 °F
K	conductivity of the liquid Btu/ft hr °F
M	defined in Eq. (24)
(dP/dz)	Pressure Gradient $lbf/ft^2/ft$
Pr	Prandtl number $\mu_\ell C_p/K$
(q/A)	heat flux $Btu/ft^2 hr$
Re_ℓ	local liquid Reynolds number $\frac{G(1-x)D}{\mu_\ell}$
S	perimeter ft
St^*	Stanton Number $\frac{h_z}{\rho_\ell c_p u_\tau}$
T	temperature °F
U	mean velocity ft/hr
u_τ	friction velocity $\sqrt{\frac{g_o \tau_o}{\rho_\ell}}$ ft/hr

v_z	local velocity in the axial direction ft/hr
W_ℓ	liquid flow rate lbm/hr
W_v	vapor flow rate lbm/hr
x	quality
y	radial distance from the wall ft
z	axial distance from the condensation starting point ft
α	void fraction
α_ℓ	thermal diffusivity $K/\rho_\ell C_p$ ft^2 hr
β	U_i/U_L
δ	thickness of the condensate film ft
ε	eddy diffusivity ft^2hr
θ	angle of inclination
μ	viscosity lbm/ft hr
ν	kinematic viscosity ft^2/hr
ρ	density lbm/ft^3
τ	shear stress lbf/ft^2
τ_v	vapor shear stress on the liquid film lbf/ft^2
Γ	liquid flow rate per unit circumference lbm/ft hr

Subscript

δ	liquid vapor interface
e	exit
f	friction
g	gravity
h	thermal
i	interface
ℓ	liquid

L	total condensing length
m	momentum
o	wall
v	vapor
z	local

APPENDIX A

Sample Calculation

Given Conditions

$G = 250{,}000$ lbm/ft^{2}hr

$T_{sat} = 86°F$

$T_o = 76°F$

physical properties (from Du Pont Table of F-22)

Viscosity	$\mu_\ell = 0.557$ lbm/hr ft
	$\mu_v = 0.0322$ lbm/hr ft
Conductivity	$K_\ell = 0.0495$ Btu/hr ft°F
Specific heat	$C_p = 0.305$ Btu/lbm °F
Latent heat	$h_{fg} = 76.470$ Btu/lbm
Density	$\rho_\ell = 73.278$ lbm/ft^3
	$\rho_v = 3.1622$ lbm/ft^3

$Pr = 3.43$

$D = 0.493$ in

Assuming that complete condensation occurs in the tube, the quality change is divided into 20 steps. A sample calculation will be done for the quality change from 72.5% to 67.5%. The local heat transfer coefficient at x = 0.7 will be considered as the average value in this quality change.

From Eq. (7)

$$\left(\frac{dP}{dz}\right)_f = -16.96 \text{ lbf/ft}^2\text{/ft}$$

From Eq. (4) with $(S/A_z) = D/4 = 0.0103$ ft, $\tau_o = 0.174$ lbf/ft^2

From Eq. (22), $u_\tau = 992$ ft/hr

Take for a first trial $D(dx/dz) = -0.001$.

From Eq. (13), $\left(\frac{dP}{dz}\right)_m = 1.36$ lbf/ft^2/ft

For a horizontal tube $\left(\frac{dP}{dz}\right)_g = 0$

From Eq. (3), $\frac{dP}{dz} = -16.96 + 1.36 = -15.60$ lbf/ft^2/ft

From Eq. (12), $\alpha = 0.95$

From Eq. (26), $Re_\ell = \frac{(1 - 0.7)(250{,}000)(0.493)}{(0.557)(12)} = 5532$

From Eq. (27c), $5532 = -256 + 12\delta^+ + 10\delta^+ \ln\delta^+$

By trial and error calculate $\delta^+ = 99.7$

Then from Fig. 3 at $\delta^+ = 99.7$, $\beta = 1.25$

From Eq. (18), $F_o = 19.20$ lbf/ft^2/ft

From Eq. (24), $M = 0.084$

From Eq. (23c), $F_2 = 34.82$

From Eq. (21b), $h_z = \frac{(73.278)(0.305)(992)}{(34.82)} = 637$ Btu/hr ft^{2}F

Since $\frac{q}{A} = h_z \Delta T = \frac{\pi}{4} \frac{D^2 \; G \; h_{fg}}{\pi D} \frac{\Delta x}{\Delta z}$

$$D\frac{\Delta x}{\Delta z} = \frac{4 \; h_z \Delta T}{G \; h_{fg}} = \frac{4(637)(10)}{(250{,}000)(76.470)} = 0.00133$$

Recalculate using this magnitude instead of 0.001. The final results $h_z = 637$, convergence is very rapid. Then

$$\Delta z = \frac{(\Delta x)(D)}{0.00133} = \frac{(0.05)(0.493)}{(0.00133)(12)} = 1.53 \text{ ft}$$

the increment of length required to change the quality from 72.5% to 67.5%.

A similar calculation should be made for each Δx of 5% to determine the corresponding h_z and Δz. A plot of h_z and x vs z may be constructed. Also P vs x or z may be plotted.

APPENDIX B

Heat Transfer Analysis

The universal velocity was assumed in the liquid layer

$$
\begin{array}{ll}
0 < \delta^+ < 5 & v_z^+ = y^+ \\
5 < \delta^+ < 30 & v_z^+ = -3.05 + 5 \ln y^+ \\
30 < y^+ & v_z^+ = 5.5 + 2.5 \ln y^+
\end{array}
\tag{B-1}
$$

where

$$v_z^+ = v_z/\sqrt{g_o \tau_o/\rho} = v_z/u_\tau ; \quad \delta^+ = \frac{\delta}{\nu}\sqrt{\frac{g_o \tau_o}{\rho}}$$

Rewrite Eq. (19a) as follows

$$\tau = \frac{\rho_\ell}{g_o}\left(1 + \frac{\varepsilon_m}{\nu_\ell}\right) u_\tau^2 \frac{dv_z^+}{dy^+} \tag{B-2}$$

Solve this for ε_m with Eq. (B-1)

$$
\left.
\begin{array}{ll}
0 < \delta^+ < 5, \ \tau \simeq \tau_o & \varepsilon_m = 0 \\
5 < \delta^+ < 30, \ \tau \simeq \tau_o & \varepsilon_m = \nu_\ell \left(\frac{y^+}{5} - 1 \right) \\
30 < \delta^+, \ \tau = F_o(\delta - y) + \tau_v \text{ and } \nu \ll \varepsilon_m & \\
\qquad \varepsilon_m = \frac{\nu_\ell}{2.5}\left[y^+ - \frac{M}{\delta^+}(y^+)^2 \right] &
\end{array}
\right\}
\tag{B-3}
$$

where

$$M \equiv \frac{F_o}{\tau_o} \frac{\delta^+ \nu}{u_\tau}$$

Rewrite Eq. (19b) in the following form assuming $q/A \simeq (q/A)_o$:

$$\frac{1}{h_z} = \frac{T_\delta - T_o}{(q/A)_o} = \int_0^{\delta^+} \frac{\nu_\ell}{\rho_\ell C_\ell (\alpha_\ell + \varepsilon_h) u_\tau} \, dy^+ \tag{B-4}$$

Taking $\varepsilon_h = \varepsilon_m$, substitute Eq. (B-3) into (B-4) and obtain Eq. (21) where F_2 is given by Eq. (23) in the three zones.

The results of this analysis can be put in an alternative form. Eq. (21) can be rewritten as follows:

$$h_z^* = \frac{Pr}{F_2} \left(\frac{\delta^+}{M}\right)^{1/3} \tag{B-5}$$

$$h_z^* = \frac{h_z}{k} \left(\frac{\nu_\ell \mu_\ell}{g_o F_o}\right)^{1/3} \tag{B-6}$$

The results can be plotted as shown in Fig. 14 and involve

$$\left.\begin{aligned} \tau_v^* &\equiv \frac{\tau_v}{F_o} \left(\frac{\nu_\ell \mu_\ell}{g_o F_o}\right)^{-1/3} \\ \delta^* &\equiv \delta \left(\frac{\nu_\ell \mu_\ell}{g_o F_o}\right)^{-1/3} \end{aligned}\right. \tag{B-7}$$

where

$$\delta^+ = \delta^* (\delta^* + \tau_v^*)^{1/2} \tag{B-8}$$

Then

$$M = \frac{1}{1 + \tau_v^*/\delta^*} \tag{B-9}$$

The momentum equation for the vapor core, Fig. 1, is

$$- \frac{dP}{dz} A_v - \tau_v S_v + \frac{a}{g_o} \rho_v A_v = \frac{1}{g_o} \frac{d}{dz} (U_v W_v) - U_i \frac{dW_v}{dz} \tag{B-10}$$

Again substituting Eq. (12) and (14) in Eq. (B-10)

$$\tau_v \frac{4}{\alpha D} = - \frac{dP}{dz} + \frac{a}{g_o} \rho_v - \frac{G^2/\rho_v}{g_o D} D \frac{dx}{dz} \left[2 \frac{x}{\alpha} + \frac{1-2x}{\alpha} \left(\frac{\rho_v}{\rho_\ell}\right)^{2/3} - \frac{\beta(1-x)}{\alpha(1-\alpha)} \left(\frac{\rho_v}{\rho_\ell}\right) \right] \tag{B-11}$$

For assumed magnitudes of δ^+, Pr and τ_v^*, calculate Re_ℓ from Eq. (28), M from Eq. (B-9), δ^* from Eq. (B-8), F_2 from Eq. (26) and h_z^* from Eq. (B-5). With these calculations, Fig. 14 can be drawn and is an alternative presentation of results.

APPENDIX C

Tables of Data

Run 1 G = 303,000 lbm/ft^2 hr T_{sat} = 86°F R-22

Measured

Sec No	T_{vapor}	$T_{o\,out}$	W_{water}	ΔT_w	dP/dz
1	83.17	71.25	2400	1.482	15.3
2	82.97	71.45	2134	1.450	24.7
3	82.40	69.47	2718	1.250	21.2
4	82.31	70.89	1807	1.531	14.1
5	82.04	70.06	1953	1.473	15.3
6	81.88	67.92	2254	1.186	9.4

Calculated

Sec No	Q/A	$T_{o\,in}$	ΔT	h	x_m
1	9200	72.73	10.44	880	93.8
2	7990	72.74	10.23	780	82.5
3	8450	70.83	11.57	730	72.2
4	7150	71.05	10.26	696	62.4
5	7450	71.26	10.78	692	53.3
6	6820	69.03	12.85	530	44.3

Heat Balance Error -8.5%

Run 2 G = 485,000 lbm/ft^2hr T_{sat} = 81°F R-22

Measured

Sec No	T_{vapor}	$T_{o\,out}$	W_{water}	ΔT_w	dP/dz
1	81.64	72.81	1690	2.61	73.2
2	81.19	70.79	2800	1.58	62.5
3	81.01	70.12	1920	2.29	57.8
4	80.19	68.84	1510	2.85	47.2
5	79.62	68.13	1730	2.50	53.0
6	79.44	67.89	1910	2.20	43.6

Calculated

Sec No	Q/A	$T_{o\,in}$	ΔT	h	x_m
1	9,440	74.33	7.31	1290	96.4
2	11,400	72.64	7.55	1330	88.3
3	11,350	71.95	9.06	1250	79.4
4	11,100	70.63	9.56	1160	70.6
5	11,200	69.93	9.69	1150	61.8
6	10,850	69.64	9.80	1100	53.3

Heat Balance Error +1.6%

Run 3 G = 250,000 lbm/ft^2hr T_{sat} = 86°F R-22

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w	dP/dz
1	86.57	70.37	3310	1.27	24.2
2	86.39	71.02	2560	1.47	20.5
3	85.96	70.12	2490	1.37	17.0
4	85.77	69.38	2910	1.07	13.0
5	85.69	69.67	2790	1.02	13.0
6	85.58	71.04	1750	1.36	7.1

Calculated

Sec No	Q/A	$T_{o\ in}$	ΔT	h	x_m
1	10,850	72.12	14.45	750	91.1
2	9,730	73.09	13.30	730	74.3
3	8,810	71.54	14.42	610	59.6
4	8,050	70.68	15.09	535	46.7
5	7,350	70.85	14.84	495	34.1
6	6,150	72.03	13.55	455	23.0

Heat Balance Error +4.65%

Run 4 G = 470,000 lbm/ft^2hr T_{sat} = 85°F R-22

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w	dP/dz
1	85.53	74.23	2660	1.77	62.3
2	85.01	75.10	1670	2.27	60.9
3	85.00	71.96	3500	1.42	57.0
4	84.54	73.92	1970	1.88	47.4
5	84.19	72.22	2470	1.54	47.2
6	83.64	71.28	3090	1.24	38.5

Calculated

Sec No	Q/A	$T_{o\ in}$	ΔT	h	x_m
1	12,200	76.19	9.34	1,300	95.1
2	9,700	76.68	8.33	1,160	85.6
3	12,800	74.02	10.98	1,165	76.0
4	9,610	75.47	9.07	1,060	66.9
5	9,840	73.81	10.38	950	59.0
6	9,900	72.87	10.77	920	51.0

Heat Balance Error +6.05%

Run 5 G = 270,000 lbm/ft^2hr T_{sat} = 92°F R-22

Measured

Sec No	T_{vapor}	$T_{o_{out}}$	W_{water}	ΔT_w	dP/dz
1	92.22	74.98	2300	1.87	18.9
2	92.07	75.30	2080	2.02	24.8
3	91.81	71.90	2770	1.69	20.1
4	91.77	71.71	2690	1.54	14.2
5	90.90	73.73	2060	1.67	14.2
6	90.94	72.03	2340	1.35	9.0

Calculated

Sec No	Q/A	$T_{o_{in}}$	ΔT	h	x_m
1	11,100	76.77	15.45	718	92.0
2	10,850	77.05	15.02	721	76.2
3	12,100	73.85	17.94	674	59.7
4	10,700	73.44	18.33	584	43.3
5	8,900	75.16	15.74	566	29.2
6	8,160	73.35	17.59	465	17.0

Heat Balance Error +4.45%

Run 6 G = 240,000 lbm/ft^2hr T_{sat} = 92°F R-22

Measured

Sec No	T_{vapor}	$T_{o_{out}}$	W_{water}	ΔT_w	dP/dz
1	93.04	80.34	2440	1.16	20.1
2	92.46	80.97	2180	1.26	21.2
3	91.98	79.72	2300	1.19	16.5
4	91.88	79.28	2310	1.15	11.8
5	91.66	78.29	2600	1.04	11.8
6	91.80	77.91	2580	1.01	7.1

Calculated

Sec No	Q/A	$T_{o_{in}}$	ΔT	h	x_m
1	7550	81.52	11.52	655	93.9
2	7100	82.10	10.36	685	82.1
3	7090	80.86	11.12	636	70.6
4	6870	80.39	11.49	599	59.4
5	7000	79.42	12.24	571	48.2
6	6740	79.00	12.80	526	37.1

Heat Balance Error -1.15%

Run 7 G = 308,000 lbm/ft^2hr T_{sat} = 92°F R-22

Measured

Sec No	T_{vapor}	$T_{o\,out}$	W_{water}	ΔT_w	dP/dz
1	98.07	80.96	2510	1.83	25.4
2	97.80	82.12	1860	2.13	23.6
3	97.37	80.06	2410	1.72	21.2
4	97.02	79.26	2800	1.69	16.5
5	96.98	78.58	2840	1.39	16.5
6	96.93	77.26	2080	1.58	11.8

Calculated

Sec No	Q/A	$T_{o\,in}$	ΔT	h	x_m
1	11,850	82.87	15.20	780	92.4
2	10,200	83.77	14.03	726	78.1
3	10,700	81.79	15.58	687	64.6
4	12,200	81.23	15.79	770	49.8
5	10,200	80.23	16.75	610	35.3
6	8,500	78.63	18.80	452	23.2

Heat Balance Error +0.9%

Run 8 G = 316,000 lbm/ft^2hr T_{sat} = 103°F R-22

Measured

Sec No	T_{vapor}	$T_{o\,out}$	W_{water}	ΔT_w	dP/dz
1	103.12	84.63	2530	2.10	15.3
2	102.91	86.73	1390	2.89	28.3
3	101.79	85.33	2480	1.58	27.1
4	101.33	83.72	3000	1.50	20.0
5	100.98	80.99	3070	1.40	20.0
6	100.98	79.66	3330	1.31	15.3

Calculated

Sec No	Q/A	$T_{o\,in}$	ΔT	h	x_m
1	13,700	86.85	16.27	842	91.2
2	10,400	88.41	14.50	717	75.8
3	10,100	86.96	14.83	680	62.6
4	11,600	87.59	15.74	737	48.7
5	11,100	82.78	18.20	610	34.1
6	11,300	81.48	19.50	610	19.8

Heat Balance Error 5.5%

Tables of Data

Run 1 G = 316,000 $lb/hrft^2$ $T_{water\ in}$ = 64.6 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	94	78.8	1510	2.72
2	93.2	77.5	1450	2.49
3	92.4	76.6	1460	2.32
4	91.9	74.1	1930	1.69
5	90.2	72.7	1835	1.46
6	89.0	71.3	2170	1.17

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	80.5	13.5	10,600	785	91.5
2	79.0	14.2	9,350	658	75.2
3	78.0	14.4	8,780	610	60.4
4	75.5	16.4	8,440	514	46.4
5	73.8	16.4	6,930	423	33.9
6	72.4	16.6	6,560	396	22.1

Heat Balance error = 2.9%

Run 2 G = 354,000 $lb/hrft^2$ $T_{water\ in}$ = 64.4 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	96.0	81.9	2070	3.09
2	95.6	80.9	1120	3.42
3	95.1	78.4	1580	2.62
4	94.6	76.0	1600	2.22
5	94.0	73.6	1590	1.72
6	93.6	73.6	1375	1.80

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	84.6	11.4	16,500	1,450	88.0
2	82.5	13.1	9,900	755	68.6
3	80.1	15.0	10,700	713	53.6
4	77.5	17.1	9,200	538	39.1
5	74.7	19.3	7,060	366	27.2
6	74.6	19.0	6,400	337	17.3

Heat balance error = 0.8%

Run 3 G = 468,000 $lb/hrft^2$ $T_{water\ in}$ = 64.4 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	97.0	84.4	2,070	3.47
2	95.8	84.0	1,120	4.06
3	95.0	79.7	1,580	2.89
4	93.9	77.0	1,600	2.42
5	93.0	75.9	1,590	2.17
6	91.0	75.4	1,475	2.02

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	87.4	9.6	18,600	1,940	81.6
2	85.9	9.9	11,800	1,190	73.0
3	81.6	13.4	11,800	880	59.9
4	78.6	15.3	9,960	652	46.3
5	77.4	15.6	8,910	572	37.5
6	76.4	14.3	7,700	548	27.8

Heat balance error = 2.7%

Run 4 G = 360,000 $lb/hrft^2$ $T_{water\ in}$ = 67.9 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	95.0	82.3	2,090	2.54
2	94.7	81.0	1,490	2.58
3	94.3	79.6	1,870	2.14
4	93.7	79.3	1,500	2.22
5	93.3	77.6	1,900	1.76
6	93.0	76.5	2,770	1.38

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	84.5	10.5	13,700	1,300	90.0
2	82.6	12.1	9,930	820	73.1
3	81.3	13.0	10,300	793	58.5
4	80.7	13.0	8,600	662	45.0
5	79.0	13.3	8,630	648	32.6
6	78.1	14.9	9,900	664	13.2

Heat balance error = 13.7%

Run 5 G = 254,000 $lb/hrft^2$ $T_{water\ in}$ = 67.9 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	93.0	79.4	2,090	2.03
2	92.1	78.6	1,490	2.10
3	91.7	76.6	1,870	1.59
4	91.4	75.4	1,500	1.48
5	91.0	75.0	1,900	1.30
6	-	-	-	-

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	81.2	11.8	11,000	932	89.0
2	79.9	13.2	8,100	613	69.8
3	77.9	13.8	7,700	558	53.7
4	76.3	15.1	5,740	380	40.2
5	76.1	14.9	6,400	428	27.9
6	-	-	-	-	-

Run 6 G = 265,000 $lb/hrft^2$ $T_{water\ in}$ = 67.9 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	99.0	81.8	2,090	2.44
2	97.8	80.1	1,490	2.38
3	97.3	77.8	1,870	1.80
4	96.9	76.8	1,500	1.74
5	96.5	75.5	1,900	1.39
6	96	74.0	2,770	1.00

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	84.0	15.0	13,200	879	87.0
2	81.6	16.2	9,150	565	65.2
3	79.2	18.1	8,700	478	47.4
4	77.9	19.0	6,750	355	32.2
5	77.6	19.9	6,830	344	19.0
6	75.2	20.8	7,160	344	3.5

Heat balance error = 1.5%

Run 7 G = 155,000 $lb/hrft^2$ $T_{water\ in}$ = 51.4°F R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	84.6	67.2	1,760	2.32
2	84.2	62.7	1,500	1.92
3	83.8	62.4	1,610	1.78
4	83.5	60.5	1,900	1.40
5	82.0	58.2	2,260	0.84
6	79.4	55.2	2,360	0.56

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	68.9	15.7	10,600	675	82.8
2	63.7	20.3	7,450	367	53.8
3	63.6	20.2	7,400	366	29.7
4	61.6	21.9	6,870	314	8.8
5	59.2	22.8	5,840	256	-
6	55.8	23.6	3,440	143	-

Heat balance error = 4.9%

Run 8 G = 445,000 $lb/hrft^2$ $T_{water\ in}$ = 51.6 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	87.8	72.6	2,670	2.82
2	86.7	68.9	2,470	2.30
3	85.7	66.7	2,300	2.18
4	85.1	66.2	1,730	2.31
5	84.5	63.4	2,050	1.78
6	84.0	60.4	2,260	1.28

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	74.9	12.9	20,150	1,560	88.6
2	71.3	15.4	14,700	955	68.7
3	68.8	16.9	13,000	768	53.0
4	67.9	17.2	10,350	602	39.8
5	65.0	19.5	9,430	484	28.8
6	61.6	22.4	7,500	335	19.3

Heat balance error = 1.9%

Run 9 G = 440,000 $lb/hrft^2$ $T_{water\ in}$ = 51.4 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	88.3	72.0	2,580	2.88
2	87.7	69.1	2,350	2.65
3	86.6	68.2	1,900	2.76
4	86.1	67.8	1,470	2.60
5	85.8	64.7	1,460	2.21
6	85.1	60.9	1,190	1.74

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	75.1	13.2	19,200	1,450	89.2
2	71.6	16.1	15,500	962	69.2
3	70.6	16.0	13,500	856	52.6
4	69.3	16.8	9,250	551	39.5
5	66.1	19.7	8,360	425	29.4
6	61.8	23.3	5,350	230	21.6

Heat balance error = 1.9%

Run 10 G = 220,000 $lb/hrft^2$ $T_{water\ in}$ = 52.2 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	102.0	73.6	1,970	3.01
2	101.7	69.2	2,230	2.50
3	101.0	66.7	2,230	2.12
4	98.8	61.6	1,860	1.47
5	-	-	-	-
6	-	-	-	-

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	76.1	25.9	15,300	590	81.6
2	71.6	30.1	14,400	478	42.8
3	68.7	32.3	12,200	378	14.5
4	62.3	36.5	7,060	194	-
5	-	-	-	-	-
6	-	-	-	-	-

Run 11 G = 272,000 $lb/hrft^2$ $T_{water\ in}$ = 75.0 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	94.5	84.7	1,750	1.94
2	94.2	84.1	1,420	1.82
3	93.9	83.2	1,510	1.66
4	93.6	80.4	1,270	1.80
5	93.0	79.3	1,915	0.81
6	91.0	78.3	1,565	0.68

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	86.1	8.4	8,780	1,045	91.8
2	85.2	9.0	6.680	743	75.1
3	84.3	9.6	6,490	676	64.6
4	81.3	12.3	5,900	482	54.6
5	80.0	13.0	4,030	310	47.1
6	78.8	12.2	2,740	275	40.7

Heat balance error = 0.4%

Run 12 G = 477,000 $lb/hrft^2$ $T_{water\ in}$ = 64.0 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	93.0	82.0	1,820	3.26
2	92.2	80.5	1,030	3.50
3	91.0	77.3	1,915	2.36
4	90.2	72.9	2,280	1.50
5	87.0	69.4	2,070	0.91
6	85.0	68.7	2,280	0.80

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	84.5	8.5	15,300	1,800	91.8
2	82.0	10.2	9,350	916	78.6
3	79.2	11.8	11,600	984	67.2
4	74.4	15.8	8,840	558	56.2
5	70.3	16.7	5,350	320	48.6
6	69.5	15.5	4,720	304	43.0

Heat balance error = 2.3%

Run 13 G = 154,000 lb/hrft^2 $T_{water\ in}$ = 63.8 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	97.0	76.9	2,800	2.04
2	96.2	73.5	2,030	1.62
3	95.7	72.6	1,590	1.67
4	95.2	71.1	1,850	1.30
5	95.0	70.0	2,390	1.03
6	90.0	68.0	1,730	0.78

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	79.3	17.7	14,700	831	75.3
2	75.0	21.2	8,850	417	35.6
3	73.7	22.0	6,840	311	9.6
4	72.1	23.1	6,220	269	-
5	71.1	23.9	6,360	266	-
6	68.6	21.4	3,480	163	-

Heat balance error = 17.2%

Run 14 G = 326,000 lb/hrft^2 $T_{water\ in}$ = 63.9 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	98.5	81.8	1,870	3.20
2	98.1	80.7	1,310	3.33
3	97.9	78.9	1,660	2.80
4	97.5	76.1	1,950	2.16
5	97.0	73.3	1,760	1.53
6	96.5	80.6	2,280	1.12

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	84.3	14.2	15,500	1,090	87.7
2	82.6	15.5	11,300	728	66.4
3	81.0	16.9	12,000	710	47.8
4	77.9	19.4	10,800	556	29.5
5	73.5	23.5	6,990	298	15.8
6	81.7	24.8	6,600	266	4.4

Heat balance error = 2.7%

Run 15 G = 425,000 $lb/hrft^2$ $T_{water\ in}$ = 64.9 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	99.0	83.9	2,240	3.20
2	98.5	81.4	2,720	2.45
3	97.8	80.0	2,120	2.60
4	97.4	77.8	2,870	2.00
5	96.5	74.7	2,140	1.67
6	96.4	72.8	2,380	1.30

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	86.9	12.1	18,600	1,540	88.7
2	84.2	14.3	17,200	1,200	66.8
3	82.3	15.5	14,200	916	47.5
4	80.2	17.2	14,800	860	29.8
5	76.2	20.3	9,210	454	15.1
6	74.1	22.3	8,000	359	4.5

Heat Balance error = 3.9%

Run 16 G = 372,000 $lb/hrft^2$ $T_{water\ in}$ = 63.5 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	96.0	79.2	2,240	2.65
2	95.4	76.5	2,720	2.07
3	94.9	76.6	2,120	2.25
4	94.4	74.3	2,870	1.67
5	93.4	70.8	2,140	1.30
6	92.5	69.7	2,380	1.02

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	81.7	14.3	15,350	1,070	89.1
2	78.9	16.5	14,550	882	68.5
3	78.6	16.3	12,300	755	50.0
4	76.3	18.1	12,400	685	32.7
5	81.9	21.5	6,900	322	19.3
6	70.7	21.8	6,270	288	14.9

Heat balance error = 1.06%

Run 17   G = 358,000 $lb/hrft^2$ $T_{water\ in}$ = 65   R-12

Measured

Sec No	T_{vapor}	$T_{o\,out}$	W_{water}	ΔT_w
1	101.0	83.3	2,240	3.10
2	100.8	81.6	2,720	2.62
3	100.1	79.2	2,120	2.43
4	99.5	75.8	2,870	1.68
5	98.8	72.5	2,140	1.31
6	98.4	71.2	2,380	1.02

Calculated

Sec No	$T_{o\,in}$	ΔT	Q/A	h	x_m
1	86.2	14.8	17,900	1,210	87.7
2	84.6	16.2	18,400	1,140	61.3
3	81.3	18.7	13,300	713	38.0
4	77.8	21.7	12,500	576	19.2
5	73.7	25.1	7,250	289	4.8
6	72.2	26.2	6,270	240	-

Heat balance error = 4%

Run 18   G = 506,000 $lb/hrft^2$ $T_{water\ in}$ 63.9   R-12

Measured

Sec No	T_{vapor}	$T_{o\,out}$	W_{water}	ΔT_w
1	91.0	79.4	2,200	2.64
2	90.3	76.7	1,970	2.24
3	89.0	74.7	2,210	1.90
4	88.1	75.0	1,880	1.98
5	87.0	73.0	1,490	1.74
6	86.0	71.4	1,340	1.50

Calculated

Sec No	$T_{o\,in}$	ΔT	Q/A	h	x_m
1	81.8	9.2	15,000	1,630	92.4
2	78.5	11.8	11,400	965	79.3
3	76.4	12.6	10,800	856	67.7
4	76.6	11.5	9,650	839	57.7
5	74.1	12.9	6,700	519	49.4
6	72.2	13.8	5,200	377	43.5

Heat balance error = 0.9%

Run 19 G = 556,000 $lb/hrft^2$ $T_{water\ in}$ = 62.9 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	88.0	77.0	2,200	2.39
2	86.9	76.3	1,970	2.34
3	85.2	74.9	2,210	2.03
4	83.9	71.3	1,880	1.51
5	81.6	68.0	1,490	1.04
6	81.2	68.1	1,340	1.00

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	79.2	8.8	13,600	1,550	94.0
2	78.2	8.7	11,900	1,370	82.1
3	76.8	8.4	11,600	1,390	71.6
4	72.5	11.4	7,350	645	62.6
5	68.6	13.0	4,010	319	57.5
6	68.7	12.7	3,460	272	54.2

Heat balance error = 22.8%

Run 20 G = 257,000 $lb/hrft^2$ $T_{water\ in}$ = 68.8 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	94.6	82.0	2,020	2.38
2	94.2	82.3	1,790	2.50
3	93.8	79.1	1,720	1.93
4	93.4	79.2	1,620	2.02
5	93.0	76.3	1,540	1.44
6	92.6	77.5	1,330	1.78

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	84.0	10.6	12,400	1,170	84.6
2	84.2	10.0	11,600	1,160	79.0
3	80.5	13.3	8,600	647	58.5
4	80.6	12.8	8,480	660	47.7
5	77.2	15.8	5,720	362	38.9
6	78.5	14.1	6,130	435	4.2

Heat balance error = 20.8%

Run 21 G = 308,000 lb/hrft^2 $T_{water\ in}$ = 68.8 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	104.2	88.7	2,020	3.38
2	103.9	90.4	1,790	3.26
3	103.6	83.8	1,720	2.81
4	103.2	83.8	1,620	2.47
5	102.8	78.0	1,540	1.78
6	102.4	78.9	1,330	2.05

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	91.7	12.5	18,500	1,480	84.0
2	92.8	11.1	15,100	1,360	55.4
3	85.8	17.8	12,500	703	25.7
4	85.5	19.7	10,700	543	14.6
5	79.1	23.7	7,090	299	3.0
6	80.0	22.4	7,050	315	-

Heat balance error = 22.8%

Run 22 G = 307,000 lb/hrft^2 $T_{water\ in}$ = 79 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	109.0	91.8	1,870	2.33
2	108.1	91.8	1,710	2.39
3	107.9	90.9	1,450	2.32
4	107.7	90.6	1,420	2.26
5	107.5	89.6	1,400	2.09
6	107.2	89.2	1,270	2.07

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	93.6	15.2	11,250	740	90.2
2	93.5	14.6	10,600	723	71.2
3	92.3	15.6	8,700	553	54.3
4	91.9	15.8	8,300	525	39.4
5	90.8	16.7	7,560	453	25.5
6	90.3	16.9	6,800	402	13.0

Heat balance error = 4.9%

Run 23 G = 314,000 $lb/hrft^2$ $T_{water\ in}$ = 79.5 R-12

Measured

Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	110.5	95.8	1,870	2.77
2	110.0	94.5	1,710	2.79
3	109.6	93.7	1,450	2.78
4	109.2	92.6	1,420	2.56
5	108.8	91.4	1,400	2.36
6	108.4	91.0	1,270	2.36

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	97.0	13.5	13,400	992	88.6
2	96.5	13.5	12,300	914	66.5
3	95.4	14.2	10,400	734	44.5
4	94.1	15.1	9,400	623	31.2
5	92.8	16.0	8,550	534	14.7
6	92.3	16.1	7,750	482	3.7

Heat balance error = 4.3%

Run 24 G = 327,000 $lb/hrft^2$ $T_{water\ in}$ = 79.1 R-12

Measured

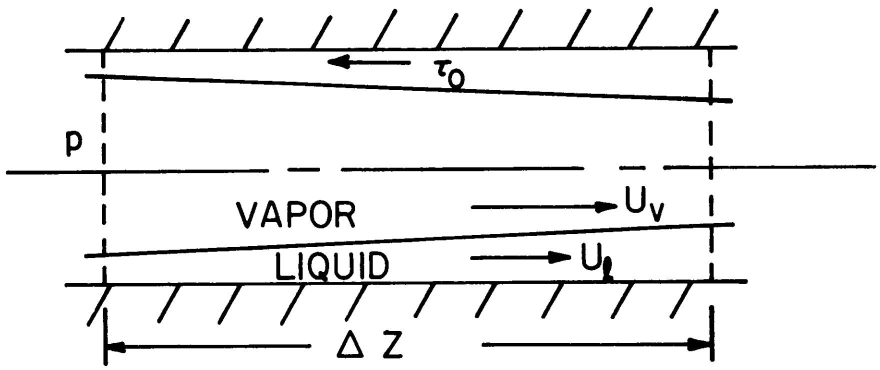
Sec No	T_{vapor}	$T_{o\ out}$	W_{water}	ΔT_w
1	118.7	97.5	1,870	3.32
2	118.4	98.2	1,710	3.57
3	118.1	96.4	1,450	3.38
4	117.8	92.9	1,420	2.69
5	117.5	88.9	1,400	2.12
6	117.1	88.9	1,270	2.00

Calculated

Sec No	$T_{o\ in}$	ΔT	Q/A	h	x_m
1	100.1	18.6	16,100	865	86.5
2	100.8	17.6	15,800	896	59.4
3	98.4	19.7	12,700	643	35.6
4	94.5	23.3	9,870	424	16.6
5	90.1	27.4	7,670	280	4.1
6	90.0	27.1	6,560	242	-

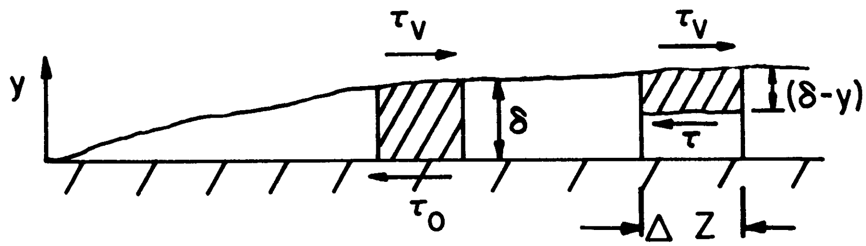
Heat balance error = 4.7

Figure Captions



τ_o
p
VAPOR
U_v
LIQUID
U_l
Δ Z

FIG 1



τ_v
τ_v
y
δ
$(\delta-y)$
τ
τ_o
Δ Z

FIG. 2

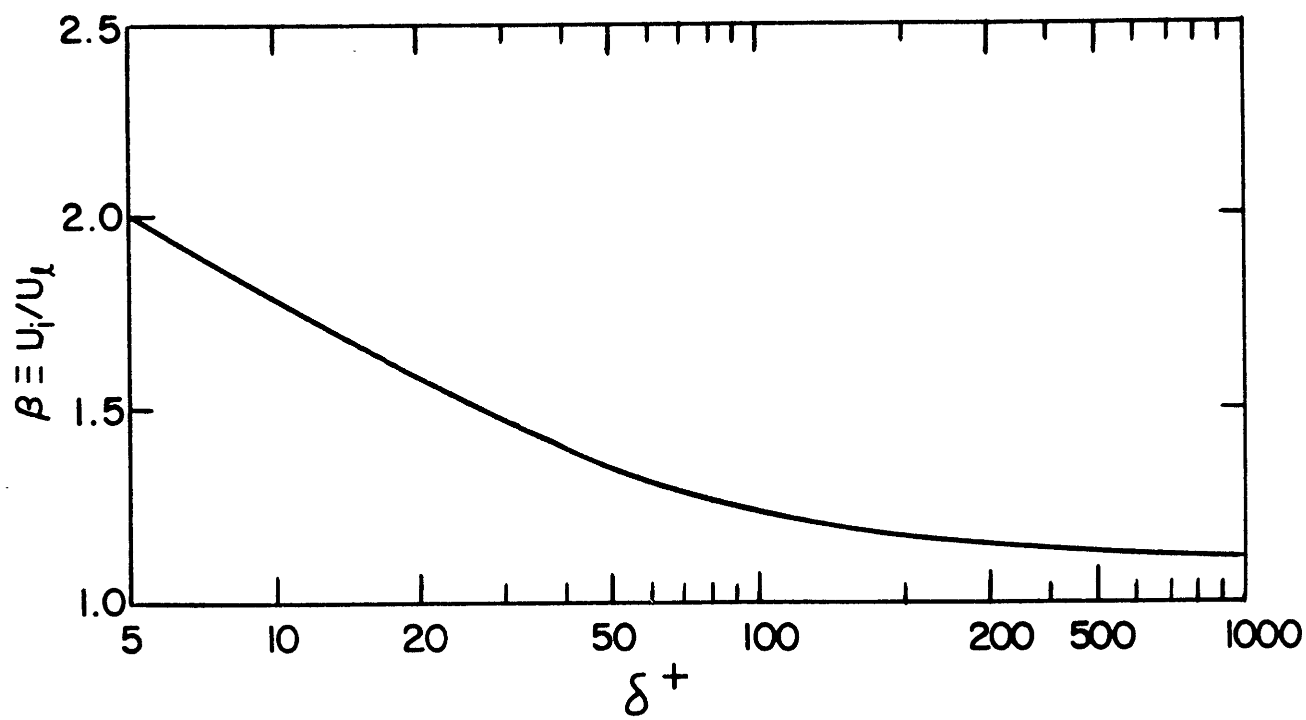


FIGURE 3 δ^+ vs. β

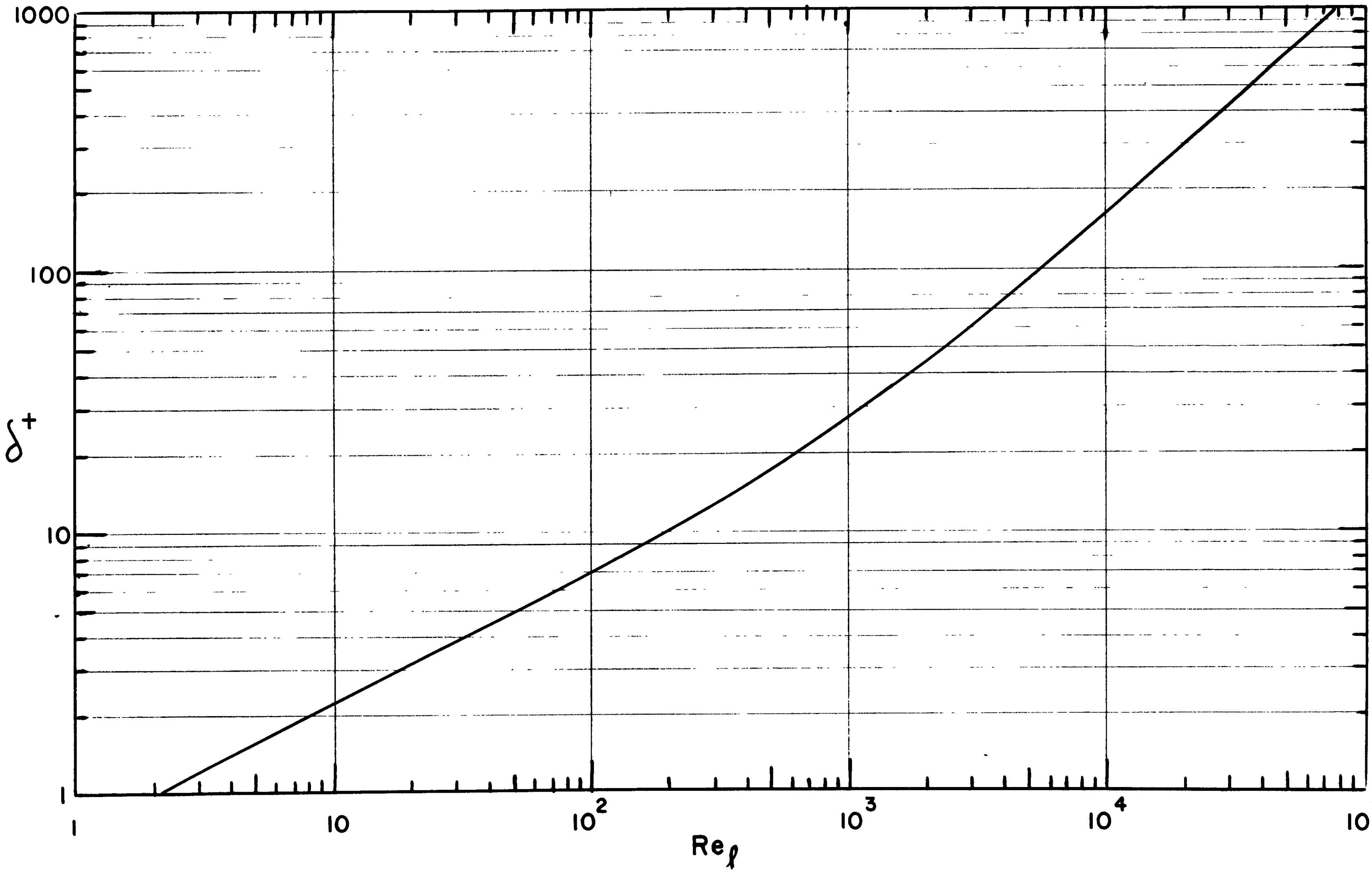


FIGURE 4 DIMENSIONLESS FILM THICKNESS δ^+

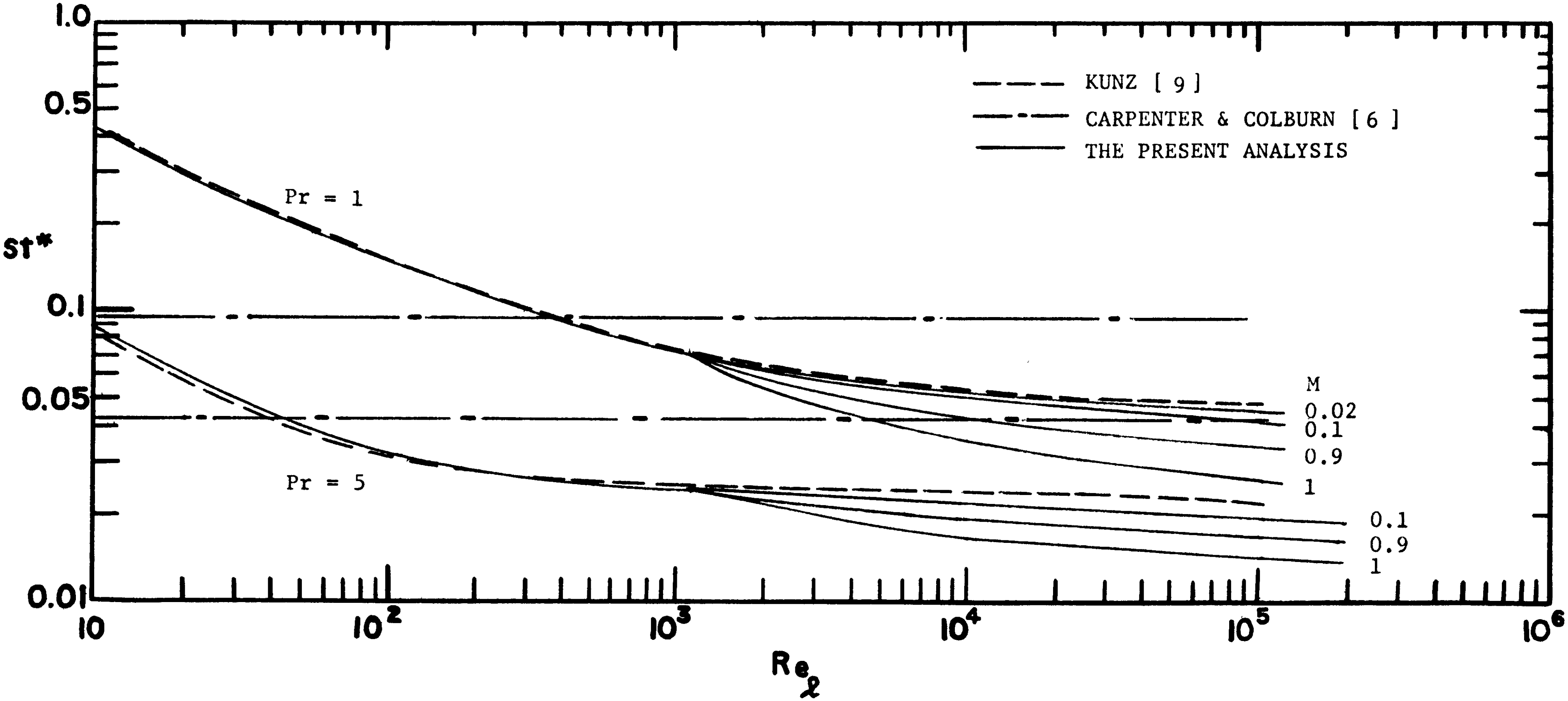

KUNZ [9]
CARPENTER & COLBURN [6]
THE PRESENT ANALYSIS
Pr = 1
Pr = 5
M
0.02
0.1
0.9
1
0.1
0.9
1
St*
1.0
0.5
0.1
0.05
0.01
10
10^2
10^3
10^4
10^5
10^6
Re_ℓ

FIG 5

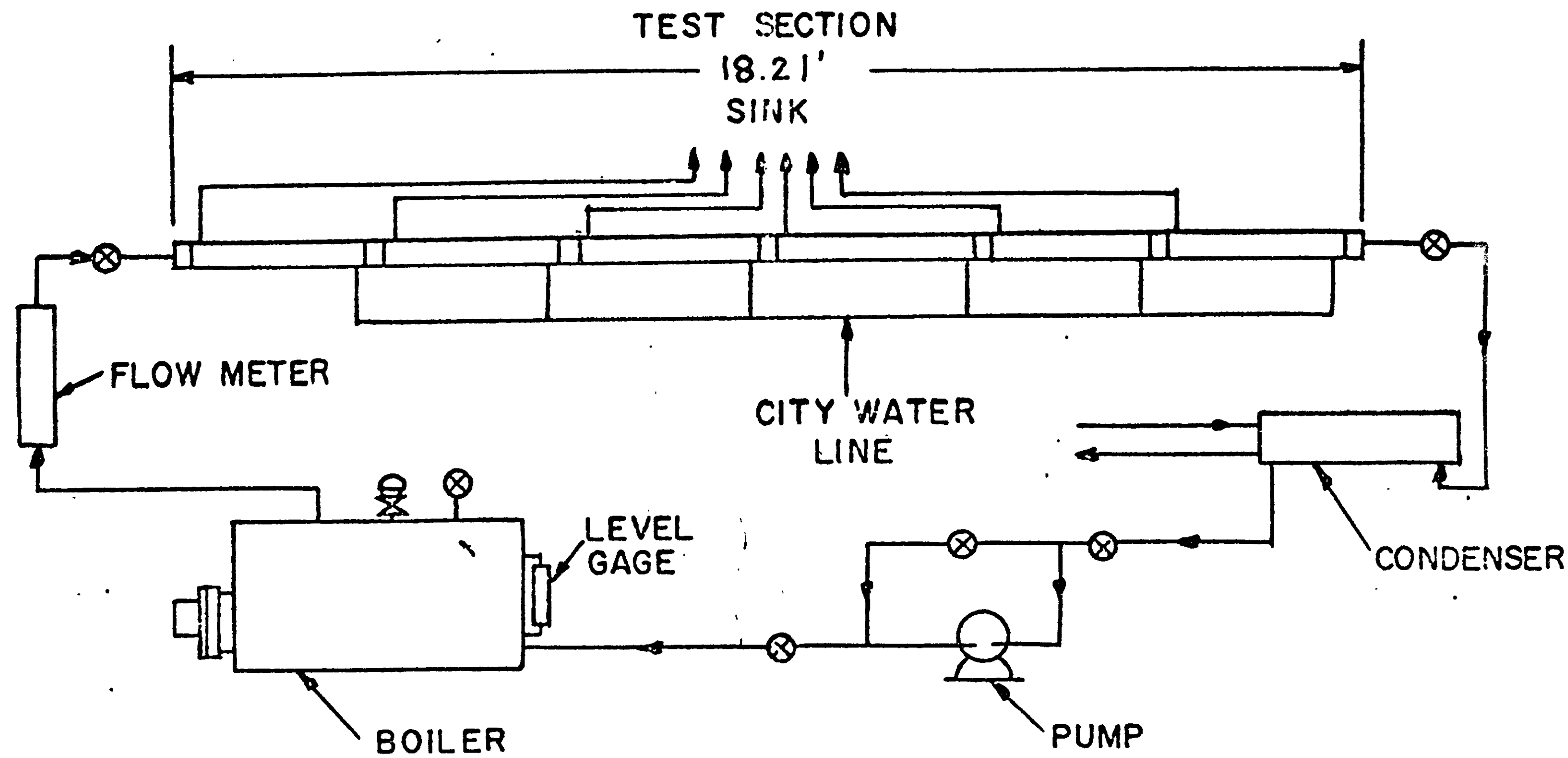


FIGURE 6 SCHEMATIC DIAGRAM OF APPARATUS

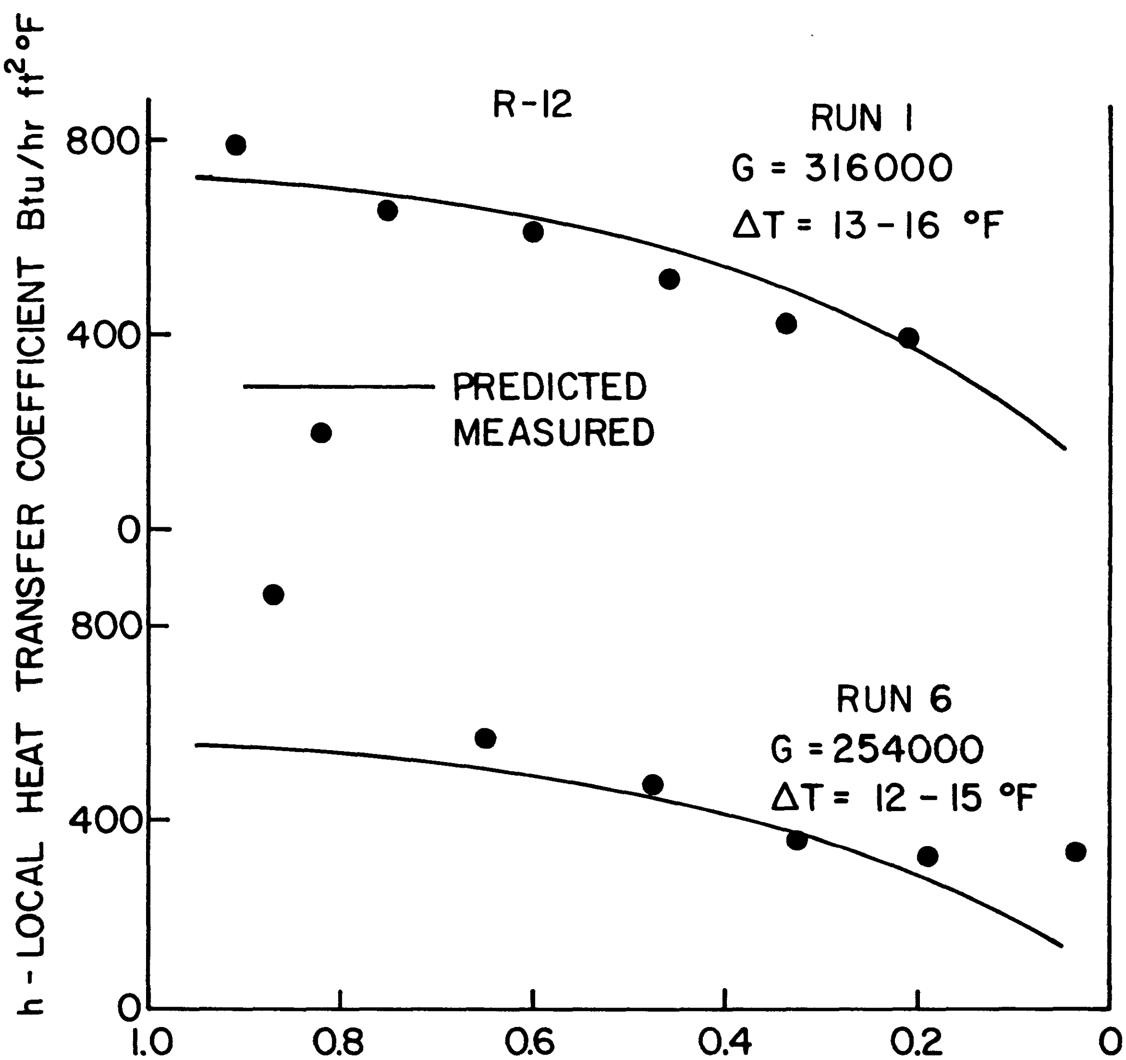


FIG.7 LOCAL HEAT TRANSFER COEFFICIENT FOR R-12 COMPARED WITH ANALYSIS

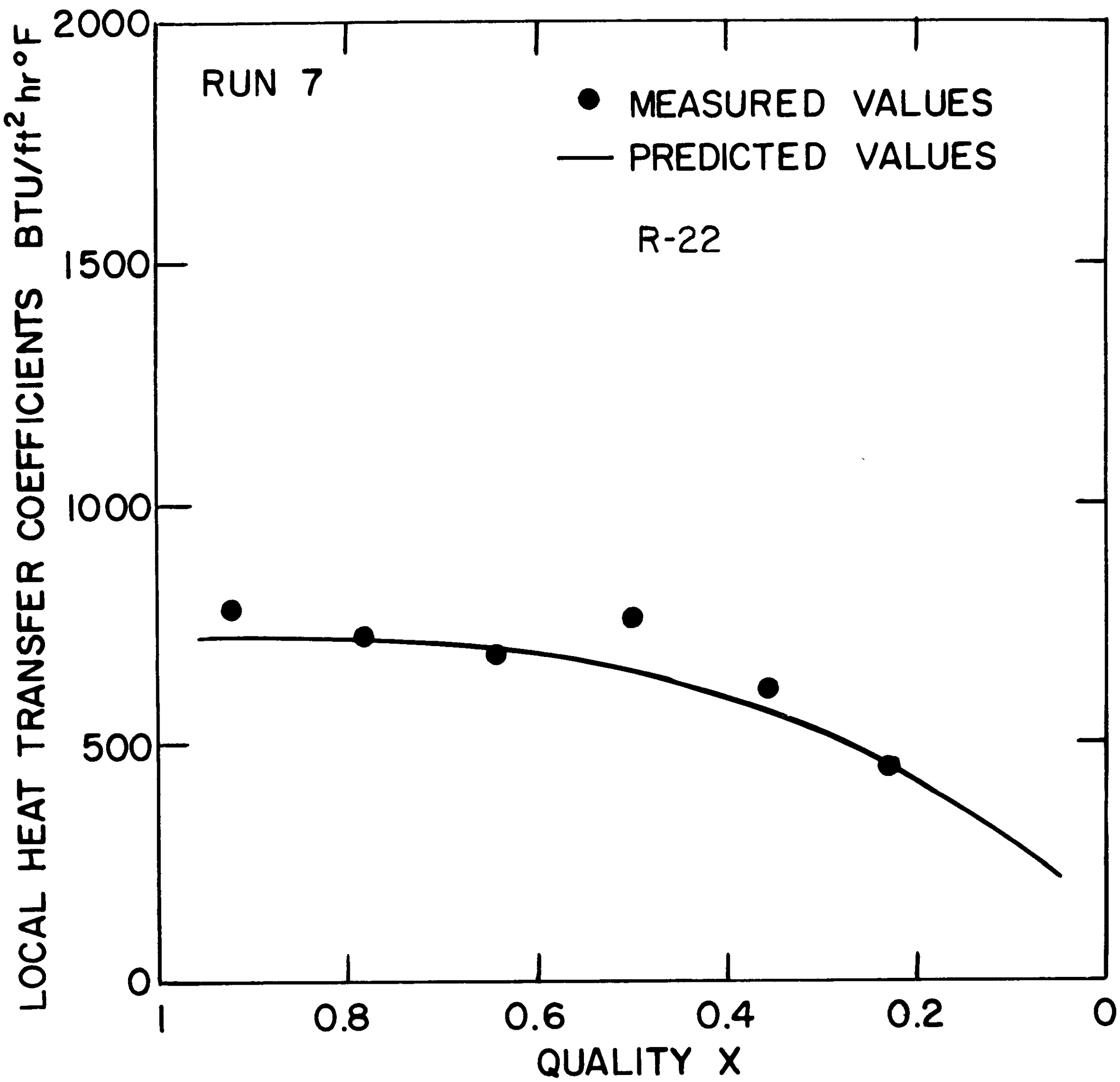


FIGURE 8 LOCAL HEAT TRANSFER COEFFICIENTS

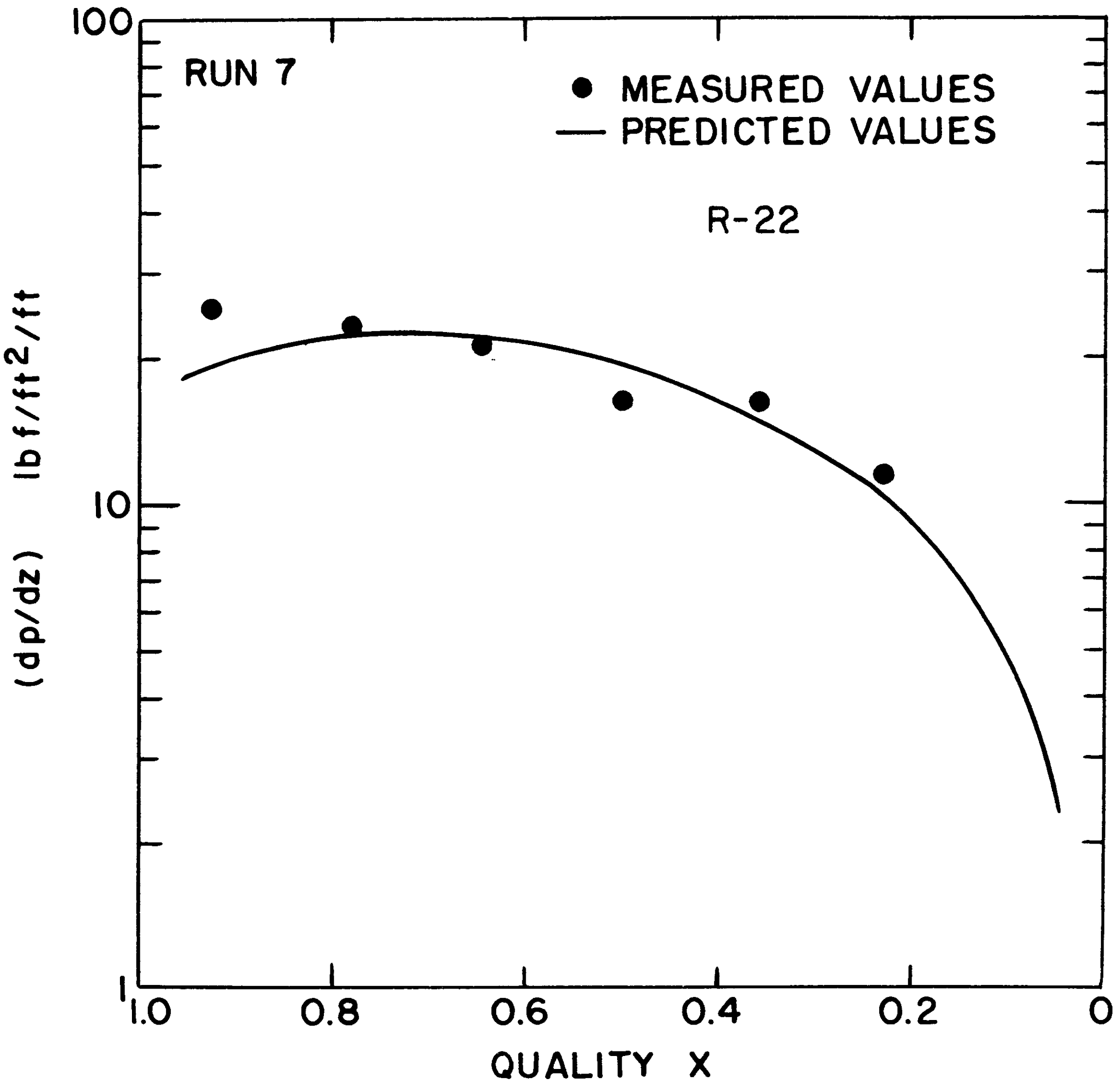
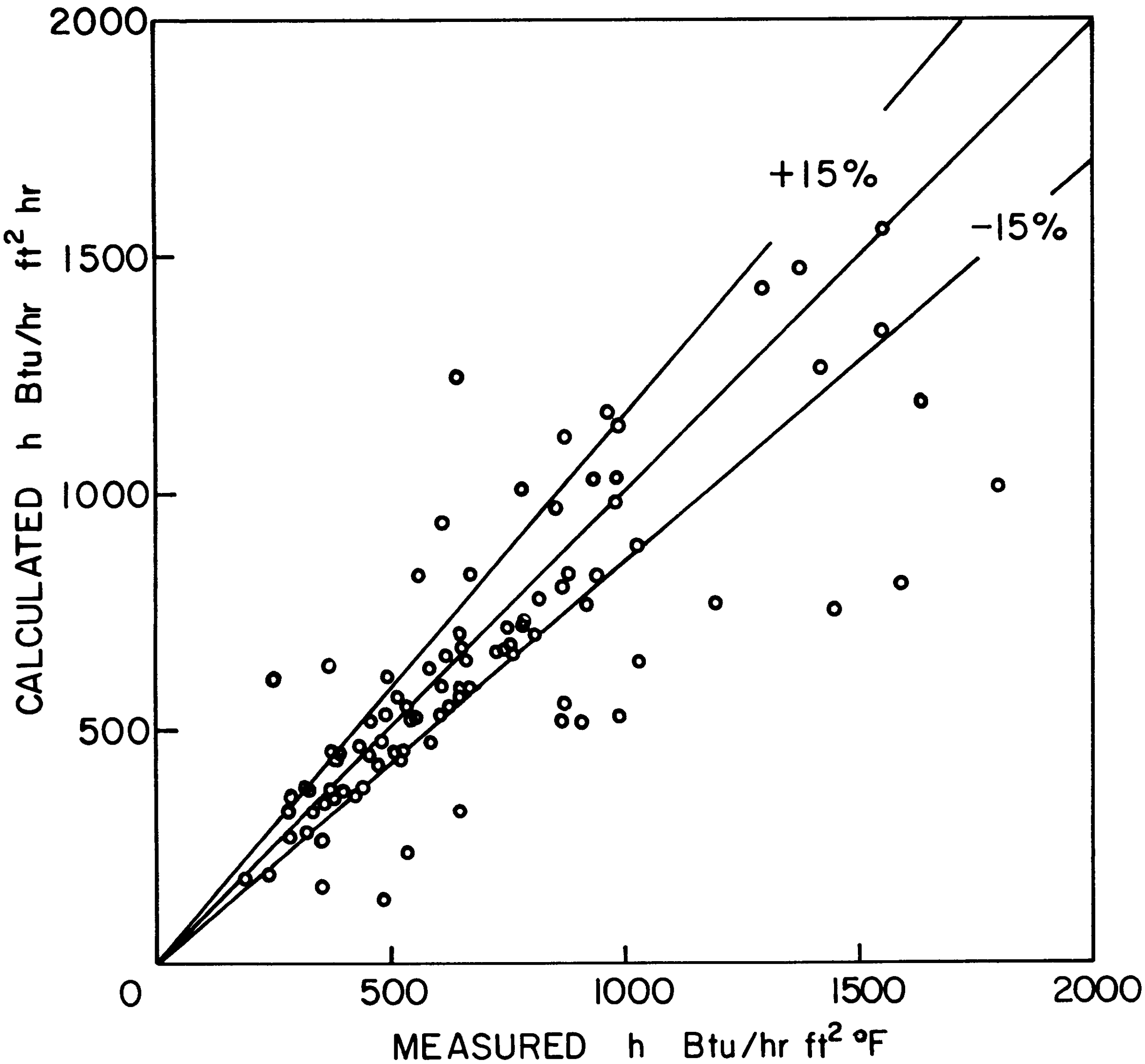


FIGURE 9 TOTAL STATIC PRESSURE GRADIENTS



R-12 HEAT TRANSFER DATA COMPARED WITH ANALYSIS

FIG. 10

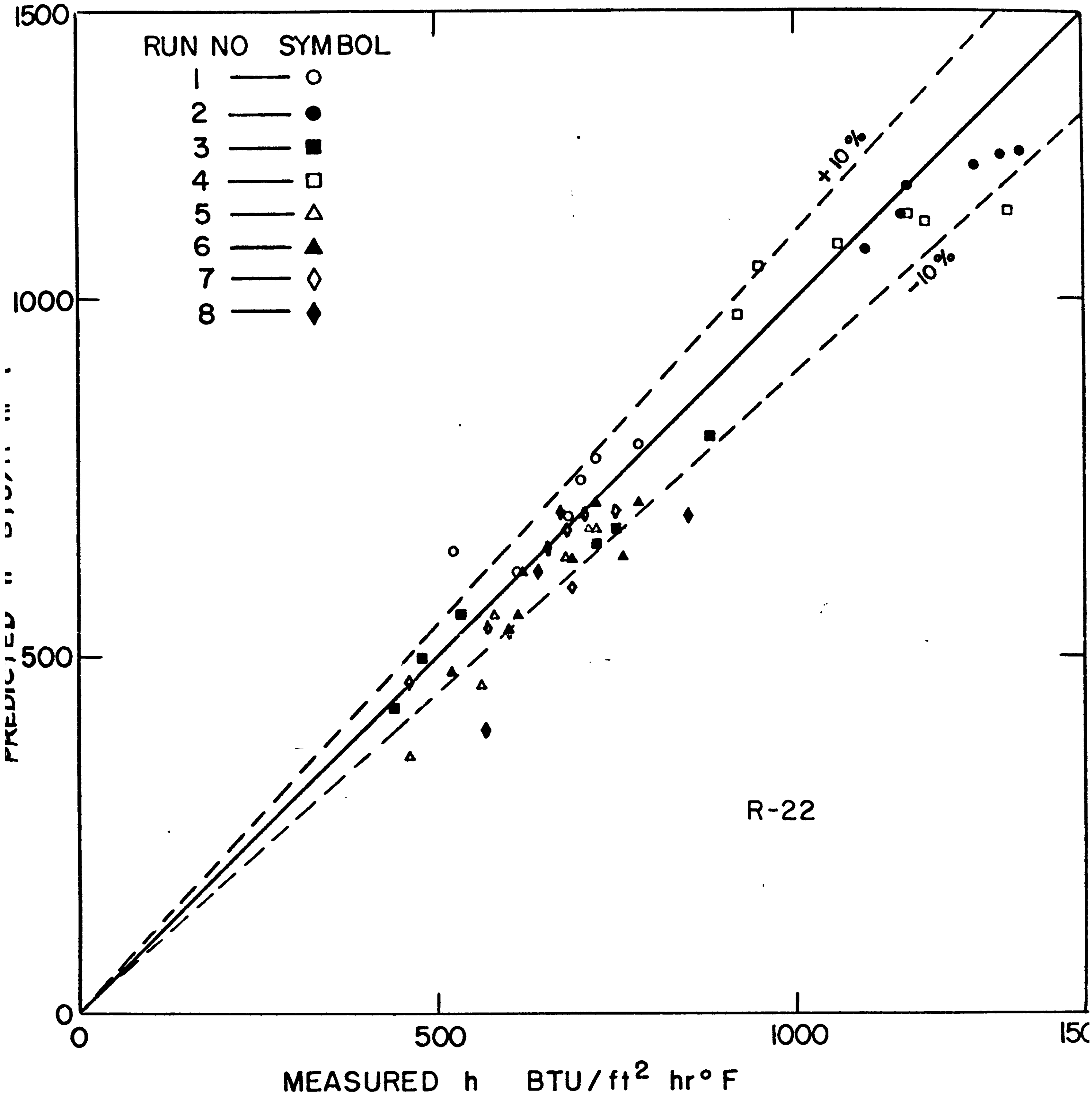
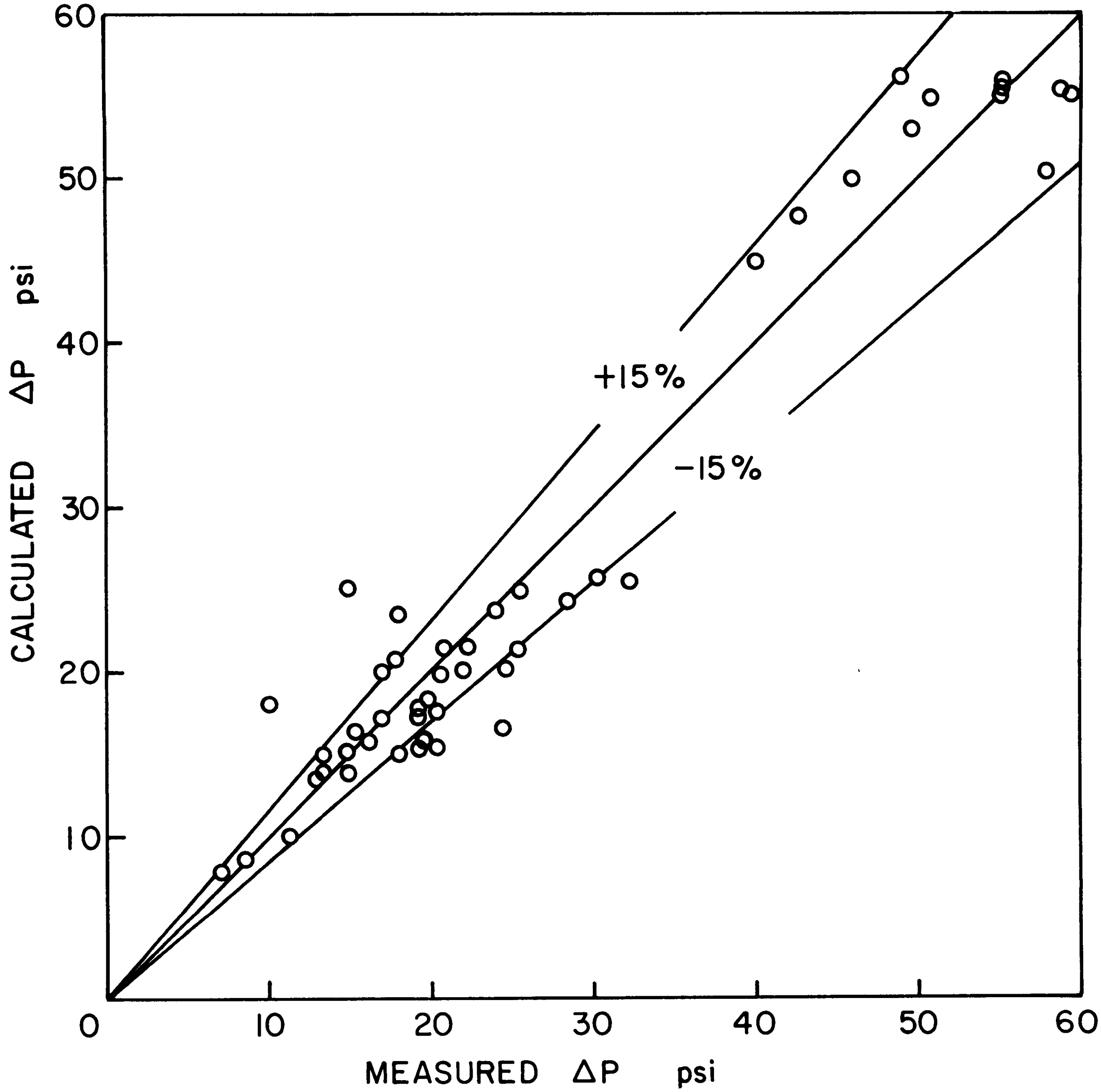


FIGURE 11 DATA COMPARED WITH PRESENT ANALYSI



R-22 PRESSURE DROP DATA COMPARED WITH ANALYSIS

FIG. 12

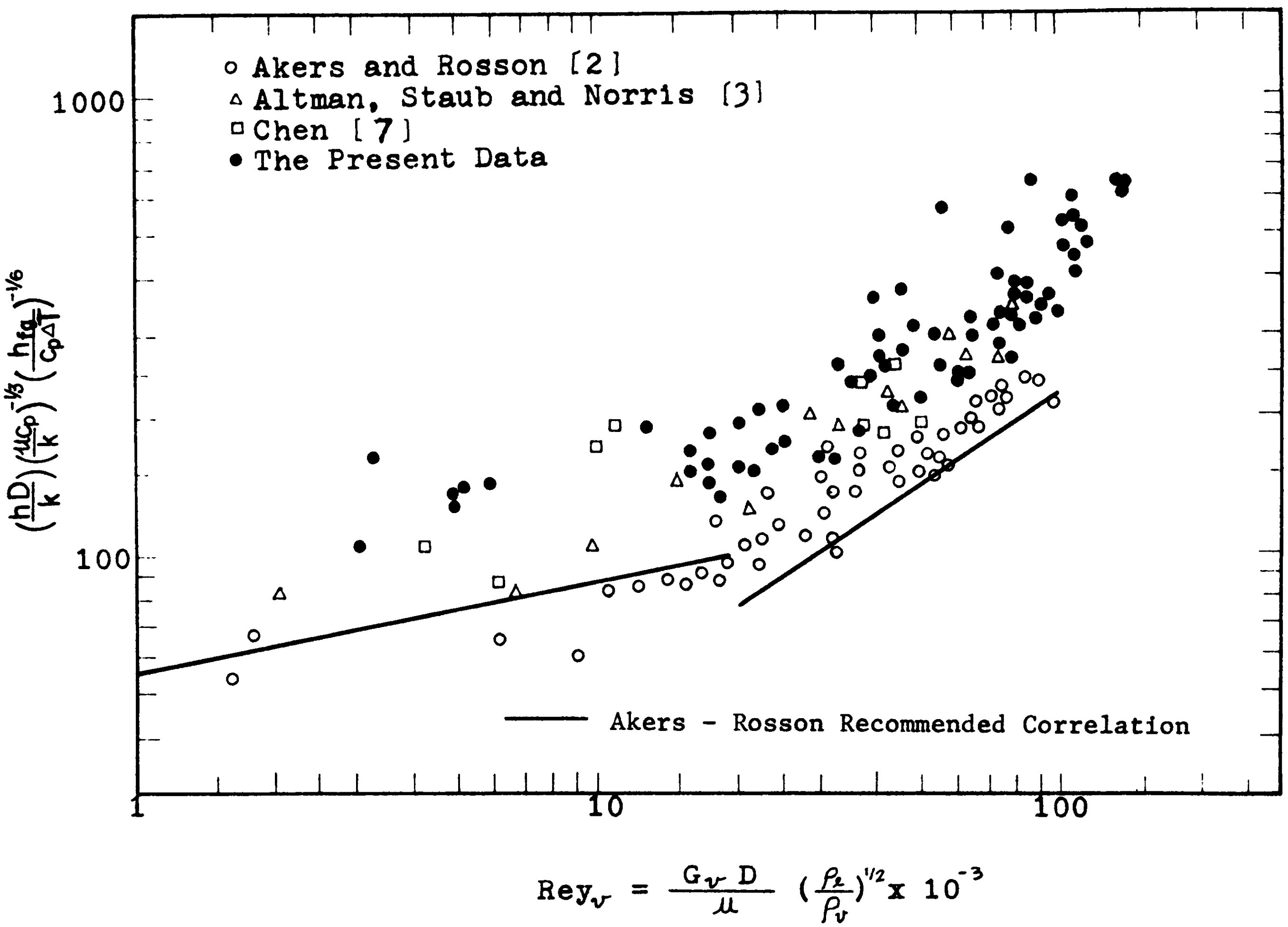


Fig. 13. Data on Akers-Rosson Plot

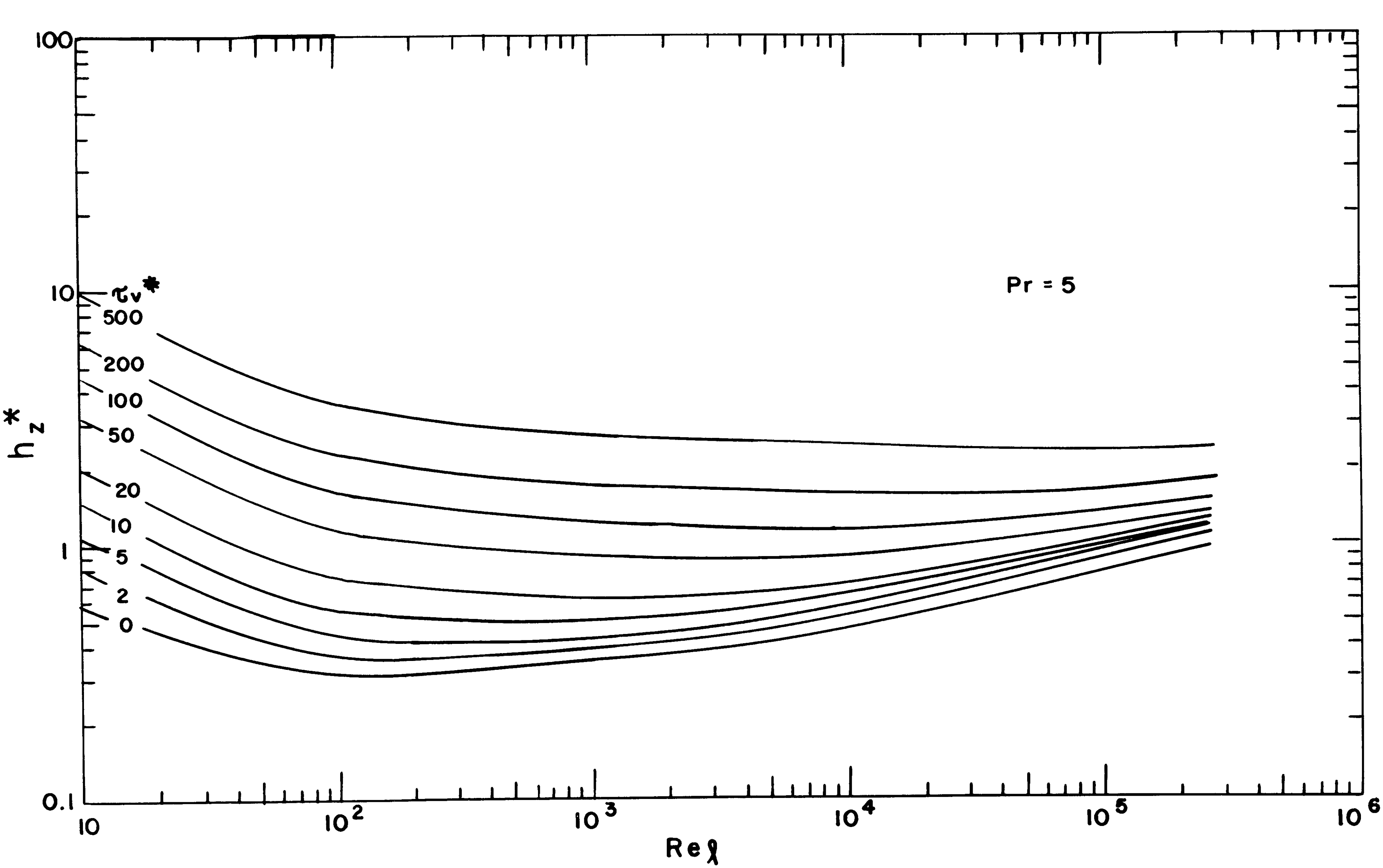


FIGURE 14 DIMENSIONLESS LOCAL HEAT TRANSFER COEFFICIENTS (Pr=5)